

Increasing the Efficiency of Greenhouse Dehumidification

Submitted to:
The Ohio State University

Submitted by:
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Abstract:

The escalation of energy costs and conservation concerns demand improvements in the applications of energy consumption. Therefore, development of improved methods to increase the efficiency of greenhouse dehumidification is necessary. In this experiment, a controlled environment dehumidification unit was designed based upon double-polyethylene greenhouses. The inside dry-bulb temperature and relative humidity were maintained near 65°F and 85%, respectively. The efficacy of the proposed design is limited to outside dry-bulb temperatures that are less than the dew-point temperature within the greenhouse. Testing of this dehumidification unit prototype will determine the accuracy of the theoretical design model and the feasibility of using a full-scale system. Accurately measuring humidity in structures is difficult, and the needed accuracy of measurements could not be obtained from this experiment to validate the theoretical model.

Project Description:**Background and Significance**

Many elements must be in proper balance to maintain a successful controlled living environment for plants. According to Walker et al., 1968, there are considerable data available concerning the effects of temperature, light, and carbon dioxide on certain plant responses, but the influences of humidity in a greenhouse environment have not been investigated thoroughly. This is likely a result of the difficulty in providing accurate measurement and control of humidity in a greenhouse (LePoidevin et al., 1981).

The effects of humidity on plant health are indeed significant. According to Jolliet, 1994, humidity affects photosynthesis, dry matter production, and the development and severity of some diseases. Fungal diseases are especially a concern since their spores are present in greenhouse air and can thrive under the right temperature and humidity. Botrytis (Gray Mold), for example, forms at a high relative humidity (>90%) and can harm most greenhouse crops (Nederhoff, 1997).

Primary factors altering the humidity in a greenhouse are transpiration, evaporation of irrigation water, ventilation, infiltration, and condensation. The former two factors are the major sources of water vapor in greenhouses. Solar radiation, leaf area index and the vapor pressure deficit substantially affect transpiration within the greenhouse (Jolliet, 1994). The next two elements depend greatly on the tightness of the greenhouse structure and the chosen method of dehumidification. Variance in humidity resulting from ventilation and infiltration is directly proportional to vapor pressure difference between the inside and outside air (Jolliet, 1994). Condensation decreases humidity in the air, but the condensate must be removed to effectively lower the overall moisture in the environment.

Some of the current methods used for dehumidification in greenhouses include simultaneous heating and ventilation, the use of desiccants, condensation, and mechanical dehumidifiers. According to LePoidevin et al., 1981, the most common method of reducing humidity in greenhouses is by simultaneously venting the moist, inside air while heating the less humid incoming air to the desired temperature. This is effective in lowering humidity levels and does not require expensive equipment. However, at cold outside temperatures the energy consumption for this method is excessive since energy not required to remove the moisture alone is lost (Jolliet, 1994).

Condensation may also be implemented as a means of dehumidification with proper removal of the condensate. This process also does not require all of the additional expenses that desiccants and commercial dehumidifiers require and may be enough to control humidity in polyethylene greenhouses. According to Jolliet, 1994, controlled dehumidification methods in single-glazed, plastic-covered greenhouses are usually not necessary due to the existing vapor pressure deficit inherent in these structures. When considering a double-glazed greenhouse, condensation may also be a viable method when the solar radiation flux is at a lower level, less than 75 W/m^2 ; however, supplemental techniques need to be developed (Jolliet, 1994).

Consideration should also be given to the effects condensation has on light and thermal transmittance. Walker et al., 1969, concluded that radiation is obstructed by heavy condensation on greenhouse walls. The significance of the radiant screening could influence the placement of intended condensing surfaces. For instance, during winter days in the Northern Hemisphere, confining the condensing surface to the north wall only would minimize the impediment of radiant energy from the sun. There will be a varying degree of resistance to transmittance depending on the thickness of the condensation, whether it is film or “drop-wise”, and the location on which it forms relative to the sun or other radiant source. Nusselt's Film Theory is used to predict the thermal resistance of developing condensation (Appendix A).

The proposed design for a Controlled-Environment Dehumidification Unit (CEDU) is to utilize an existing double-polyethylene greenhouse structure with modifications made to improve water extraction and uniform air circulation between the polyethylene layers. It is believed that circulating the greenhouse air between the two layers of polyethylene film will improve the efficiency of dehumidification during cold weather by consuming less energy than traditional heat-and-vent methods.

Procedures and Equipment:

The experiment consisted of two parts, a theoretical, computer-based model and a laboratory prototype to measure actual parameters. The first part of the experiment involved development of a theoretical model that could predict the behavior of the proposed design. Input variables included values for inside and outside temperatures, air velocities of the warm and cold air streams, and the relative humidity of the warm, inside air before and after it has passed through the CEDU. Values for these variables came from collected experimental data (Appendix B). Implementation of a series of interrelated thermodynamic and heat transfer equations returned expected values for the rate of heat transfer and the mass flow rate of water for a given system. The model was done using Microsoft Excel and a personal computer. The equations used for this model and their references are given in Appendix A.

In order to determine the amount of water that needed to be removed, it was first necessary to determine an appropriate evapo-transpiration rate. This was done using the Penman-Monteith model for evapo-transpiration (Prenger, et al., 2002). From the literature review, it was decided that humidity control by condensation would be most beneficial during the evenings of winter months. Therefore, the solar flux into the greenhouse was assumed zero when calculating evapo-transpiration, and winter conditions were implicit when analyzing heat transfer coefficients.

The rate of condensation is directly proportional to the rate at which energy is transferred from the inside greenhouse air to the outside environment. Factors affecting this rate of

heat transfer will be the heat transfer coefficients of the inside and outside airstreams, which change according to air stream velocity, the thermal resistivity of the polyethylene film, the thermal resistance of the developing condensate film layer and the temperature differential across these boundary layers. Therefore, the rate of condensation for the condensing surface will vary according to the velocity, temperature and humidity of the air entering the condenser as well as the velocity and temperature of the outside air stream.

The second part of the study included building a prototype and modeling experimental conditions. The prototype for this experiment utilized a common, household refrigerator to simulate outside climatic conditions (Figure 1). The refrigerator door was replaced with a model wall section of a double-polyethylene greenhouse. The model had a wooden frame with two layers of 28"x 36" 6-mil polyethylene sealed to either side separated by a six-inch air gap. The magnetic seal from the door was removed and placed on the model frame to minimize leaks while making the inside of the refrigerator conveniently accessible for instrumentation or any alterations required during the experiment. An intake manifold was made from a piece of two-inch poly-vinyl-chloride tube for more even distribution of the incoming air. Holes were drilled into the manifold totaling an area equivalent to two-thirds of the intake area of the pipe. A gap was provided at the top of the model frame for exhausting air and a gutter was attached to the bottom of the condensing surface to transport condensate to a measuring beaker.

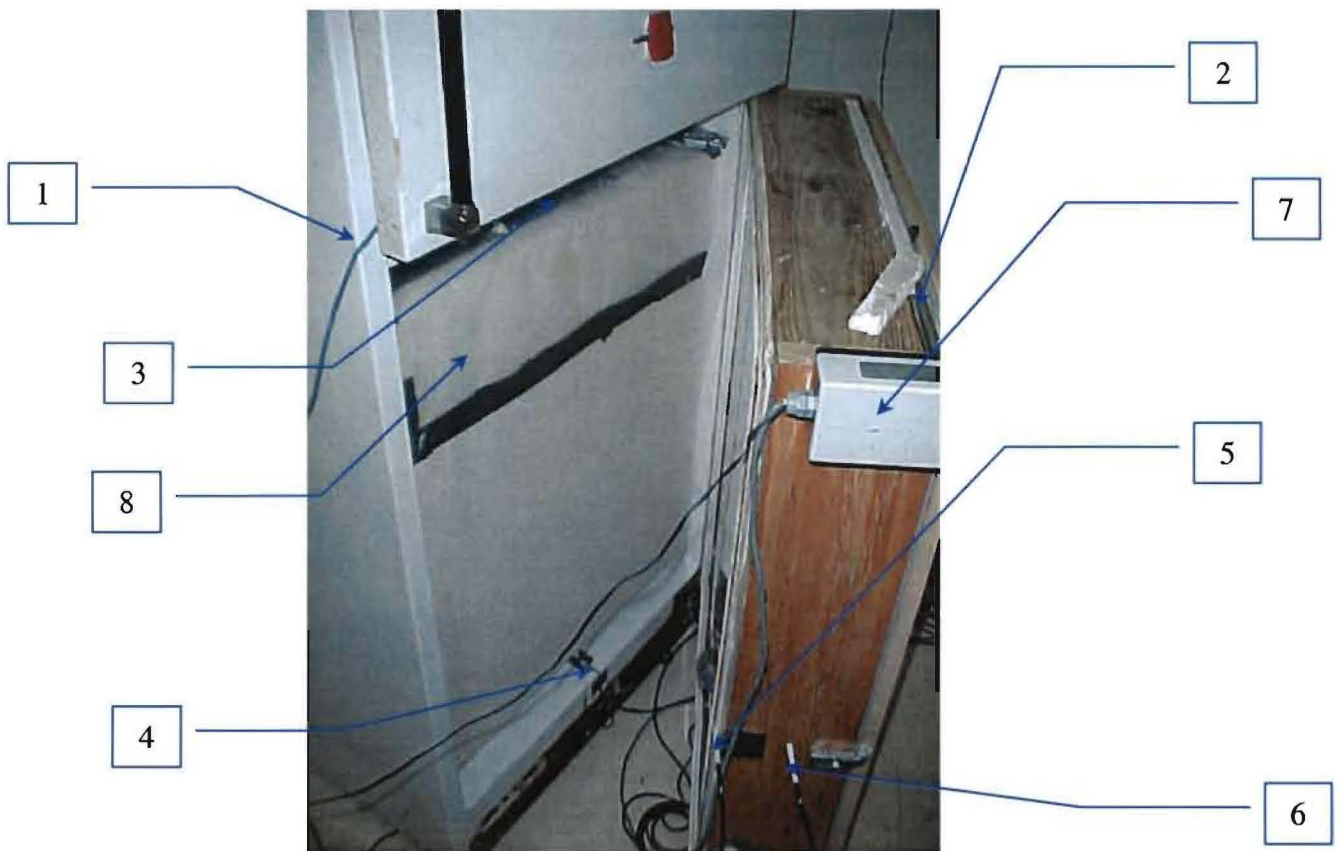


Figure 1: CEDU prototype

¹cooling coil temp., ²warm exhaust temp., ³cold inlet temp., ⁴cold exhaust temp., ⁵cold air velocity,
⁶warm air velocity, ⁷warm exhaust RH, ⁸flow restrictor

Use of the plant growth chamber (Figure 8), located in the Food, Agricultural and Biological Engineering building at the Ohio Agricultural Research and Development Center (OARDC) in Wooster, Ohio, provided a controlled environment for the experiment. The unit's proportional integral derivative (PID) controls allowed a reasonably precise temperature of 65 [°F] and a relative humidity of 85%. These parameters were also measured throughout the experiment to verify the system's accuracy and precision.

The experiment required a minimum of four temperature measurements, two relative humidity measurements and two air velocity measurements. Thermocouples and relative humidity sensors were located at the intake and exhaust of the warm, moist air stream.

Thermocouples were also placed at the intake and exhaust sides of the cold air stream.

Air velocity transducers were positioned in the middle of the warm and cold air streams to record the average velocity of the passing air. In addition to the previous measurements, thermocouples were placed in the ambient air of the growth chamber and directly on the refrigerator's cooling coil in order to monitor the stability of these temperatures.

Two fans were used to force the air for the warm and cold stream. Originally, two blowers were used in anticipation of achieving a higher range of velocity control from their higher displacements. However, possibly due to air stagnation, these higher displacements were not beneficial compared to smaller 14 and 6-Watt fans. The larger blowers also generated more heat, which warmed both air streams and altered the relative humidity of the intake to that of the ambient.

Results and Discussion:

The experimental and theoretical results for the mass flow rate of water from the CEDU are tabulated below in Table 1. Several of the calculated flow rates, notably those prior to the fifth trial, are negative. A negative flow rate of water indicates that water is being added to the system rather than removed; however, this is certainly not the case as the experimental results in each trial resulted in water removal, and thus a positive flow rate.

All of the negative values for the theoretical flow rate occurred in experiments two through four, in which a polystyrene flow restrictor was installed inside of the refrigerator to model a counter-flow heat exchanger. Therefore, it is highly probable that the flow restrictor plays a factor in the invalid theoretical results.

Table 1: Theoretical and experimental results for the mass flow rate of water from the CEDU.

	<i>Experimental Conditions</i>	<i>Avg. mdotw [lbw/hr] (theoretical)</i>	<i>Experimental Mass Flow Rate [lbw/hr] (experimental)</i>	Explanation of experimental conditions:
EXP 1-1	w/insulation w/o manifold w/ Defrost	0.046900	0.022700	w/ insulation: refers to the use of the flow restrictor (Appendix C_#8)
EXP 1-2	Fridge setting: 4	0.029900	0.013000	
EXP 1-3		0.018200	0.015900	
EXP 1-4		0.102300	0.015159	
EXP 1-5		0.032570	0.010305	w/o insulation: flow restrictor is removed
EXP 1-6		0.019197	0.011220	
EXP 1-7		0.195195	0.018818	w/ manifold: refers to the use of the PVC manifold for more even air distribution (Appendix C_#16)
EXP 1-8		0.130428	0.017198	
EXP 2-1	w/insulation w/ manifold w/ Defrost	- 0.034428	0.021143	w/o manifold: no manifold used at the inlet
EXP 2-2	Fridge setting: 4	0.025602	0.017343	
EXP 3-1	w/insulation w/ manifold, w/o defrost	- 0.034676	0.015936	w/ defrost: experiments conducted across major temperature fluctuations in refrigerator's cooling coils
EXP 3-2	Fridge setting: 4	- 0.026464	0.019308	
EXP 4-1	w/insulation, w/ manifold, w/o Defrost	- 0.002566	0.022404	w/o defrost: attempt to conduct experiments between major temperature fluctuations in refrigerator's cooling coils
EXP 4-2	Fridge setting: 4	- 0.025664	0.019239	
EXP 5-1	w/o insulation, w/manifold, w/o Defrost	0.028863	0.030823	Fridge setting: #: The temperature setting on the refrigerator
EXP 5-2	Fridge setting: 4	0.033593	0.031839	
EXP 5-3		0.023614	0.033556	
EXP 6-1	w/o insulation, w/ manifold, w/o Defrost	0.030374	0.032788	
EXP 6-2	Fridge setting: 9	0.044228	0.059674	
EXP 7	w/o insulation, w/o manifold, w/o Defrost	0.022516	0.031650	
	Fridge setting: 9			
EXP 8	w/o insulation, w/o manifold, w/o Defrost	0.123179	0.031965	
	Fridge setting: 4			
EXP 9	w/o insulation, w/o manifold, w/o Defrost	0.118961	0.023975	
	Fridge setting: 4			

Cold stream temperature recordings indicate that the temperature differential increased during trials without the flow restrictor. It is likely that this increased temperature differential may have been what was necessary to move the humidity recordings beyond the margin of error of the relative humidity transducers. When comparing the warm stream intake and exhaust on the psychrometric chart, it becomes obvious that this experiment requires very accurate measuring instruments. The absolute humidities

between the intake and exhaust are so close together that even a small error in relative humidity could result in a negative flow rate.

Another possibility for inaccuracy in recorded data may have resulted from the refrigerator's defrost cycle. As seen from the raw data in Appendix B, large spikes occurred in most of the measurements. A thermocouple was placed on the refrigerator's cooling coils and it was found that these coils could exceed temperatures of 100 °F during each defrost. Attempts were made to conduct experiments between these spikes in temperature; however, the inside temperature of the refrigerator was still variable. Depending on the relative lag-time between instruments, this is also a potential source of error.

The results indicate that the most water removed from the air occurred during trials of experiments 5, 6, 7, 8 & 9 (Table 1), in which the polystyrene flow restrictor was removed from the refrigerator. Figures 2 and 3 compare experiments done with the flow restrictor to those done without. An average of 0.017414 [lbw/hr] more was obtained from trials without the flow restrictor. The air in the former experiments may have been stagnated by a pressure build-up caused by the flow restrictor. This does not appear to be the case, however, as there is no significant difference among the cold-flow air velocities among the various trials. A more likely reason for the increase in the mass flow rate of water from the unit is due to increased turbulence and adiabatic mixing of the air within the refrigerator. A decrease in temperatures at the bottom of the cold side of the condensing wall supports this theory.

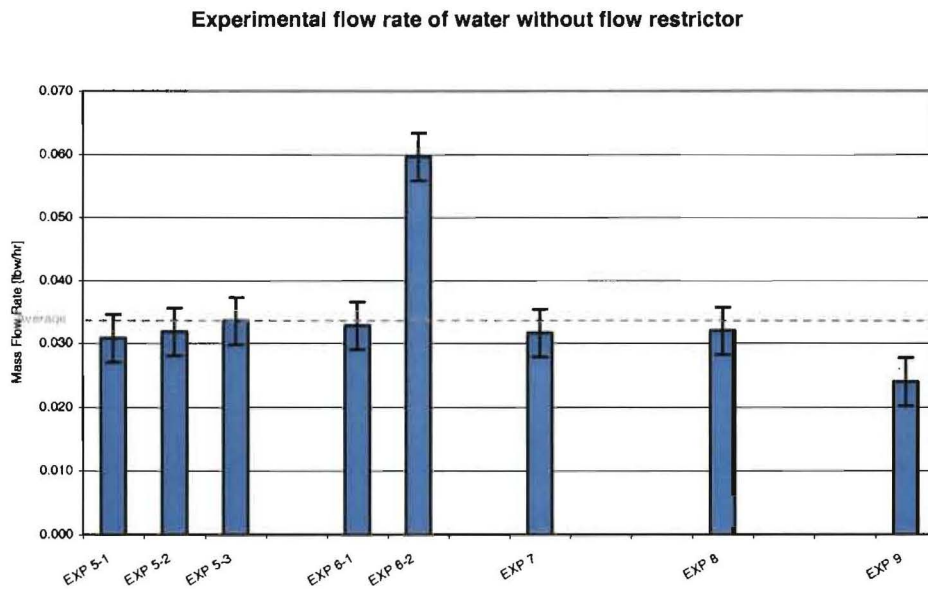


Figure 2: Experiments 5 through 9 (Unrestricted)

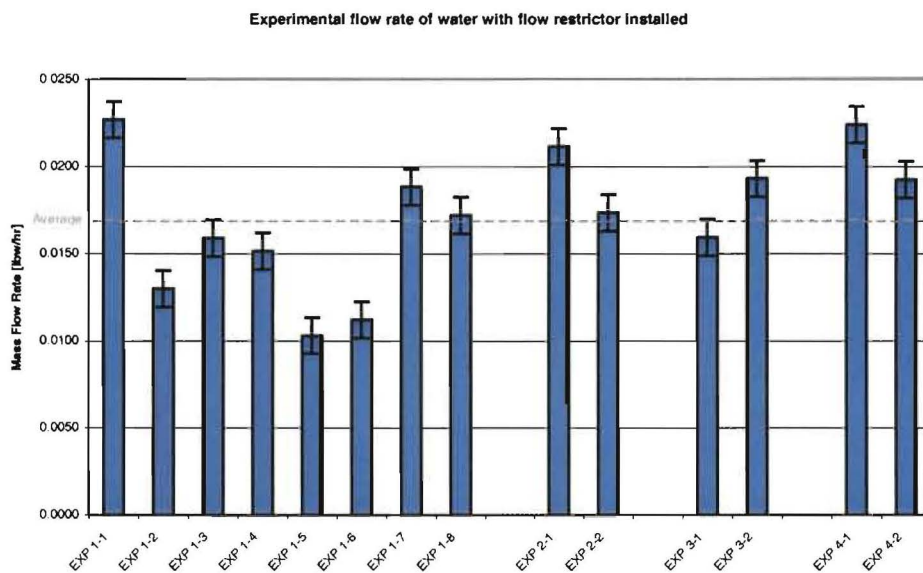


Figure 3: Experiments 1 through 4 (Flow restrictor)

The removal of the flow restrictor is a more accurate portrayal of real world conditions. The outside surface of a full-scale model will most likely be exposed to unrestricted, turbulent winds; although, it is possible that the greenhouse would be near another building, for which flow would most likely be restricted in the fashion of a cross-flow

heat exchanger. Nonetheless, for the experiments that did not model a counter-flow heat exchanger by which airflow rate could be estimated, the heat transfer equation for counter-flow heat exchangers cannot be considered valid. It is still possible to estimate the rate of heat transfer with the recorded data, and so it was done in the same manner as for a vertical wall using methodology from ASHRAE 2001.

Recommendations:

The results of this experiment could not determine the accuracy of the theoretical model; however, further research may be able to do so. Possibilities for the success of future research include more accurate measurements of relative humidity, more extreme temperature differences between the warm and cold air, or both.

Once the accuracy of the model is validated, it will give an estimate of the amount of the amount of moisture that can be removed and energy required to remove the moisture under the given conditions of the prototype. Using the model developed for the prototype, the size of a full-scale unit may be estimated based upon the removal rate of moisture needed to maintain the desired humidity level for a particular greenhouse. Figure 4 exhibits the estimated time of year in Wooster, Ohio when dehumidification by the proposed method of condensation may be used. Because the inside greenhouse air conditions are assumed to remain constant, this method is most limited by the outside dry-bulb temperature. In order for condensation to occur at any rate, the condensing surface temperature must be less than the dew-point temperature of the inside air. For a double-glazed polyethylene greenhouse, the outside temperature required to achieve

condensation was calculated to be about 36.8 °F. As Figure 4 depicts for Wooster, Ohio, a sufficient dew-point temperature will most likely be attained, and render the proposed method effective for double-glazed greenhouses, from the first week in December until late February, when the temperature is likely to be less than 37 °F.

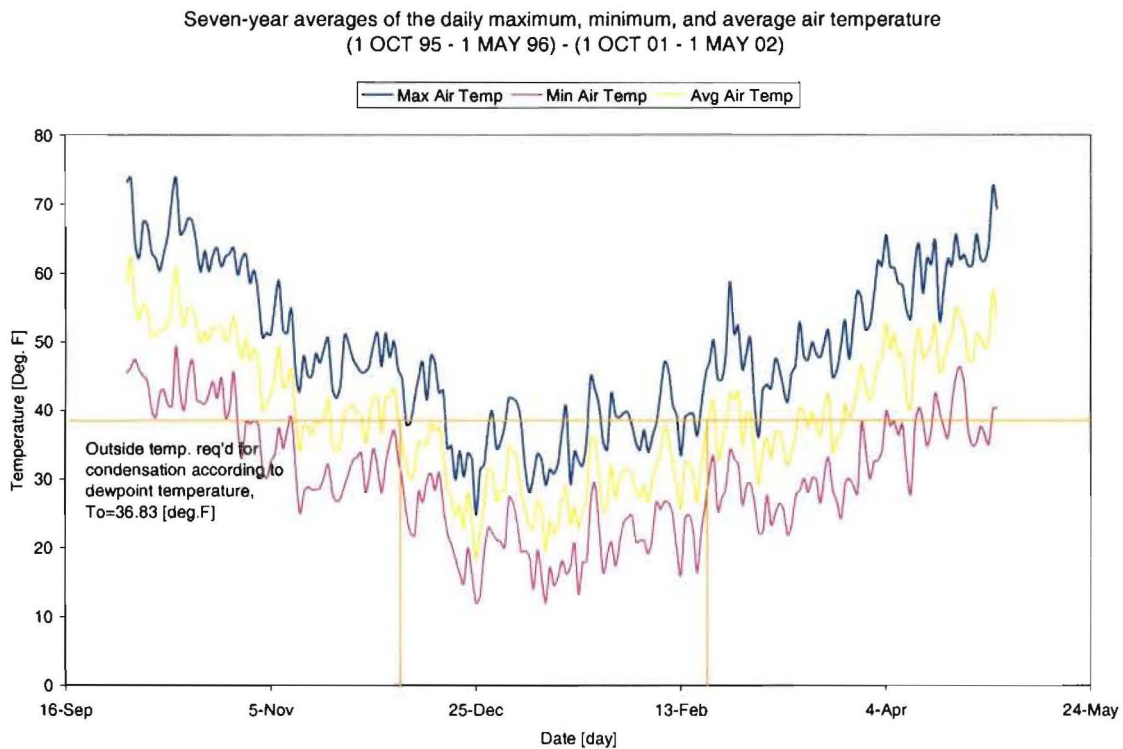


Figure 4: Timeframe for Wooster, Ohio when outside temperature is sufficient for condensation to occur using proposed method.

Appendix A

Equations used to develop theoretical model

$$Q_{cv}[\text{BTU/hr}] = \frac{A_w}{R_{TOT}} (T_i - T_{o,i})$$

$$Q_{cv} [\text{BTU/hr}] = \frac{A_w}{R_{TOT}} \frac{(T_e - T_{o,i}) - (T_e - T_{o,i})}{\ln[(T_e - T_{o,i}) - (T_e - T_{o,i})]} \quad \{\text{HX calc}\}$$

A_w = area of condensing surface [ft²]

R_{TOT} = overall thermal resistance $\left[\frac{\text{ft}^2 \cdot \text{°F} \cdot \text{hr}}{\text{BTU} \cdot \text{in}} \right]$

T_i = dehumidifier inlet temperature [°F]

T_e = dehumidifier inlet temperature [°F]

$$Q_{cv}[\text{BTU/hr}] = \dot{m}_a [(h_{ae} - h_{ai}) - \omega_i h_{gi} + \omega_e h_{ge} + (\omega_i - \omega_e) h_{fe}] * 3600$$

\dot{m}_a = mass flow rate of air [lb_{air}/sec]

{Moran, 2000; 12.51, p.656}

h_{ae} = specific enthalpy for dry air, exhaust [BTU/lbda]

h_{ai} = specific enthalpy for dry air, intake [BTU/lbda]

h_{ge} = specific enthalpy, saturated vapor, exhaust [BTU/lb]

h_{gi} = specific enthalpy, saturated vapor, intake [BTU/lb]

h_{fe} = specific enthalpy, saturated liquid, exhaust [BTU/lb]

ω_i = inlet humidity ratio [lb_w/lb_{da}]

ω_e = exhaust humidity ratio [lb_w/lb_{da}]

$$\dot{m}_w [\text{lb}_w / \text{hr}] = \dot{m}_a (\omega_i - \omega_e) * 3600$$

{Moran, 2000; 12.48, p.656}

\dot{m}_a = mass flow rate of air [lb_{air}/sec]

{ASHRAE 2001, (43), F6.17}

ω_i = inlet humidity ratio

ω_e = exhaust humidity ratio

Wall temperature, t_w [°F]:

$$t_w = 0.17 * (5.88235 * T_{o,i} + \frac{65 - T_{o,i}}{0.17 + 2/v_i^{0.78}})$$

Nusselt's number, Nu:

$$Nu = 0.943 * \left(\frac{\Delta h * (\rho_w - \rho_v) * g * y^3}{((t_{dp} - t_w) * v_w * k)} \right)^{0.25} \quad \{\text{Incropera, 10.31, p.521}\}$$

y = wall height [ft]

g = gravitational constant [ft/hr²]

t_w = wall temperature [°F]

t_{dp} = dew point temperature [°F]

k = thermal conductivity of water [BTU/hr-ft-°F]

Δh = latent heat of vaporization [BTU/lb]

ρ_w = density of water [lb/ft³]

ρ_v = density of water vapor [lb/ft³]

v_w = kinematic viscosity of water [ft²/hr]

Heat transfer coefficient for condensing film layer, h_m :

$$h_m = \frac{1.2 * Nu * k}{y}$$

y = wall height [ft]

k = thermal conductivity of water [BTU/hr-ft-°F]

Nu = Nusselt's # [Dimensionless]

R_m [ft²*F*hr/Btu]:

$$R_m = 1/h_m$$

Thermal resistance of inside air boundary layer, R_i [ft²*F*hr/Btu]:

If inside air velocity, $v_i > 16$ [fps],

$$R_i = 2 / (v_i^{0.78}) \quad \{ASHRAE 2001, (18), F3.14\}$$

Else,

$$R_i = 0.99 + 0.21 * v_i \quad \{ASHRAE 2001, (19), F3.14\}$$

Thermal resistance of outside air boundary layer, R_o [ft²*F*hr/Btu]:

If outside air velocity, $v_o > 16$ [fps],

$$R_o = 2 / (v_o^{0.78}) \quad \{ASHRAE 2001, (18), F3.14\}$$

Else,

$$R_o = 0.99 + 0.21 * v_o \quad \{ASHRAE 2001, (19), F3.14\}$$

R_{tot} overall thermal resistance [ft²*F*hr/Btu]:

$$R_{tot} = R_m + R_{poly} + R_i + R_o$$

$$R_{poly} = 0 \quad \{T.Short, pers. comm.\}$$

Specific enthalpy for saturated water vapor, h_{ge} [BTU/lb_w]:

$$h_{ge} = 1061 + 0.444 * T_e \quad \{ASHRAE 2001, (31), F6.13\}$$

T_e = exhaust temperature [°F]

Specific enthalpy for dry air, h_{ae} [BTU/lb_{da}]:

$$h_{ae} = 0.24 * T_e \quad \{ASHRAE 2001, (31), F6.13\}$$

T_e = exhaust temperature [°F]

h_{fe} :

$$h_{fe} = \text{IF}(T_e = 32, -0.02, \text{IF}(T_e = 33, 0.99, \text{IF}(T_e > 51, T_e - 31.93, \text{IF}(T_e > 40, T_e - 31.96, \text{IF}(T_e > 38, T_e - 31.97, T_e > 36, T_e - 31.98, \text{IF}(T_e > 33, T_e - 32, "Error"))))))))$$

T_e = exhaust temperature [°F]

Exhaust humidity ratio, ω_e [lb_w/lb_{da}]:

$$\omega_e = \frac{0.62198 * P_{we}}{P - P_{we}} \quad \{\text{ASHRAE 2001, (22), F6.12}\}$$

P = static pressure inside wall [psi]

P_{we} = water vapor partial pressure (exhaust) [psia]

Saturation pressure (exhaust), P_{wse} [psia]: {ASHRAE 2001, (6), F6.2}

$$P_{wse} = \exp^{(-10440.397/(T_e + 459.67)) + (-1.129465 * 10 + (-0.027022355 * (T_e + 459.67)) + 0.00001289036 * (T_e + 459.67)^2 + (-0.000000002478068 * (T_e + 459.67)^3) + 6.5459673 * \ln(T_e + 459.67))}$$

T_e = exhaust temperature [°F]

Water vapor partial pressure (exhaust), P_{we} [psia]:

$$P_{we} = RH_e * P_{wse} \quad \{\text{ASHRAE 2001, (24), F6.13}\}$$

RH_e = relative humidity of exhaust [%]

P_{wse} = saturation pressure (exhaust) [psia]

Partial pressure of dry air (exhaust), P_{ae} [psia]:

$$P_{ae} = P - P_{we} \quad \{\text{ASHRAE 2001, F6.12}\}$$

P = static pressure inside wall [psi]

P_{we} = water vapor partial pressure (exhaust) [psia]

Find: h_{ai} , h_{gi} , ω_e , P_{wsi} , P_{wi} , & P_{ai}

Repeat above equations for intake by substituting intake temperature and relative humidity in lieu of that for the exhaust.

Dew point temperature, T_{dp} [°F]:

$$T_{dp} = 100.45 + 33.193 * \alpha + 2.319 * \alpha^2 + 0.17074 * \alpha^3 + 1.2063 * \alpha^{0.1984}$$

$$\alpha = \ln(P_{wi})$$

{ASHRAE 2001, (37), F6.14}

Mass flow rate of dry air, \dot{m}_a [lbda/sec]:

$$\dot{m}_a = \frac{A_i * v_i}{((\bar{R} / M_{da}) * ((T_i + 459.67) / (P_{ai} * 144)))} \quad \{\text{Moran, 2000}\}$$

A_w = area of intake [ft²]

v_i = inside air velocity [fps]

T_i = intake temperature [°F]

P_{ai} = Partial pressure of dry air (intake) [psia]

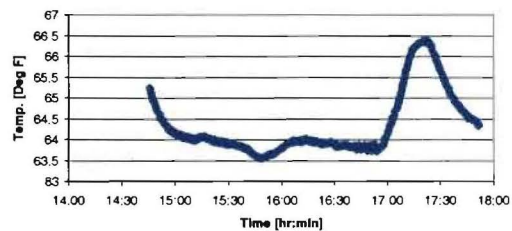
R_{bar} = universal gas constant, 1545 [ft-lbf / lbmol-°R]

M_{da} = molecular weight of dry air, 28.97 [lb/lbmol]

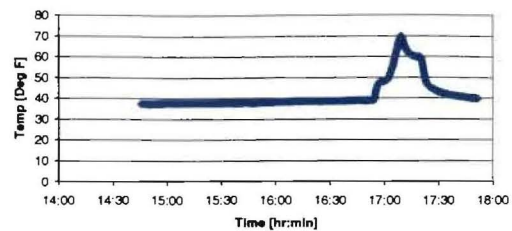
Appendix B

Graphical form of raw experimental data for select trials

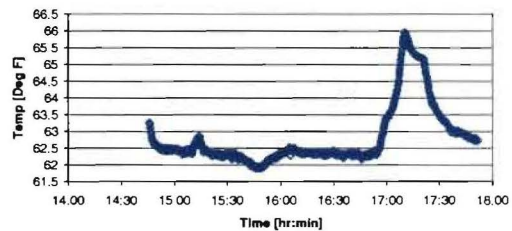
Inside Temp. (Intake)



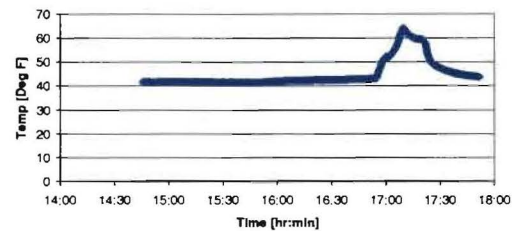
Outside Temp. (HX In)



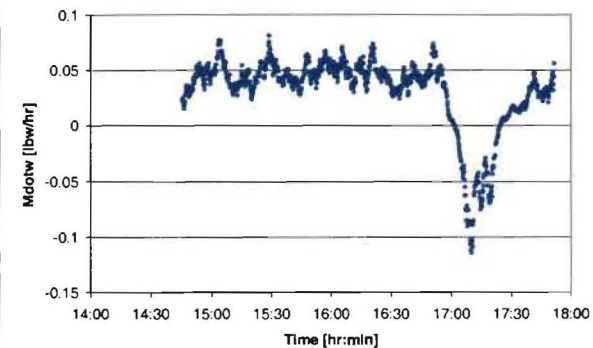
Inside Temp. (exhaust)



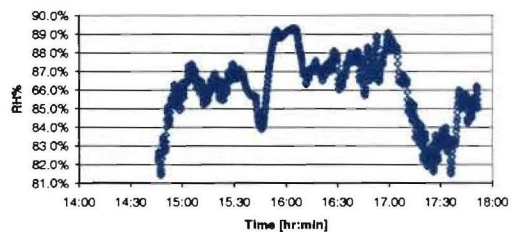
Outside Temp. (HX out)



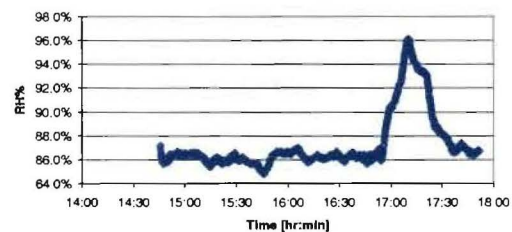
Mass Flow of Water, 2 min. Avg.



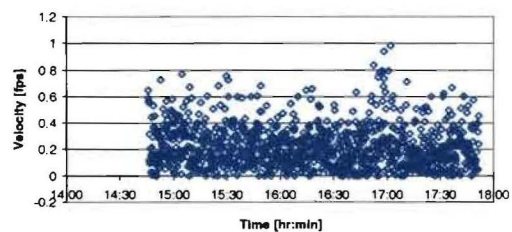
Relative Humidity (In)



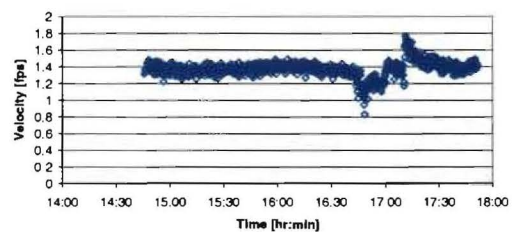
Relative Humidity (Out)



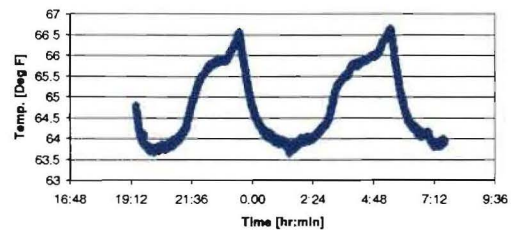
Inside Air Velocity



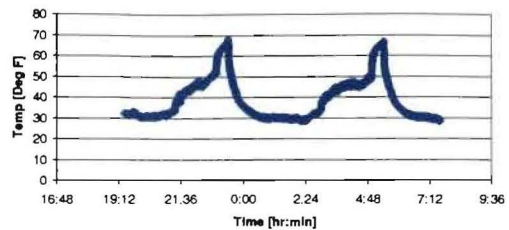
Outside Air Velocity



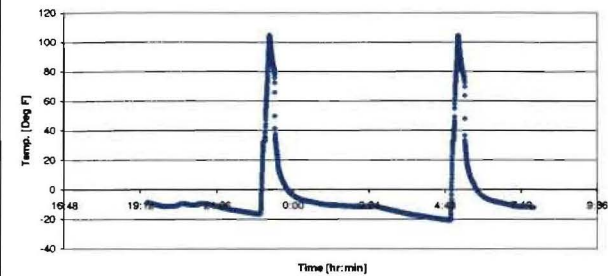
Inside Temp. (Intake)



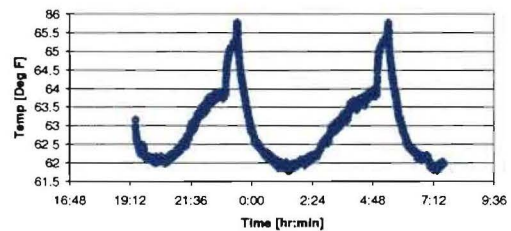
Outside Temp. (HX in)



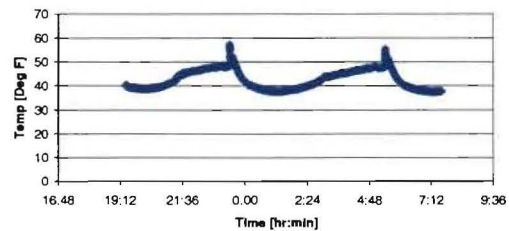
Fluctuations in cooling coil temperature



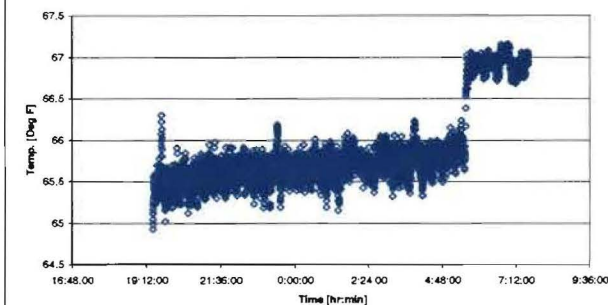
Inside Temp. (exhaust)



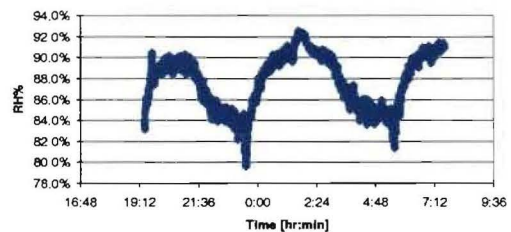
Outside Temp. (HX out)



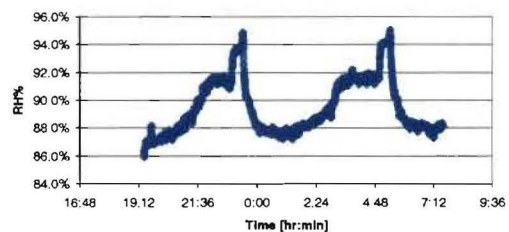
Growth chamber ambient temperature



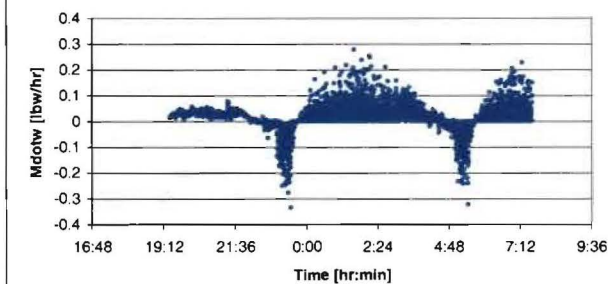
Relative Humidity (In)



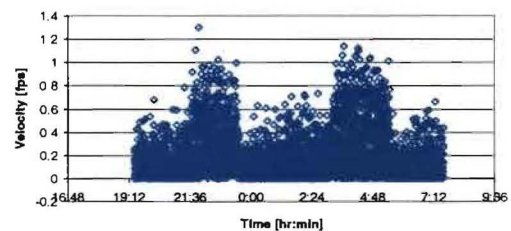
Relative Humidity (Out)



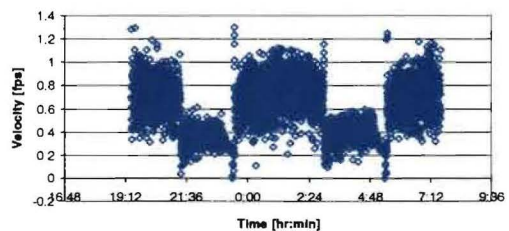
Mass Flow of Water, 2 min. Avg.



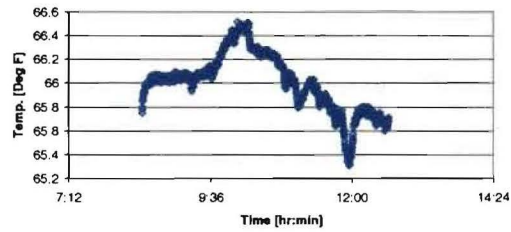
Inside Air Velocity



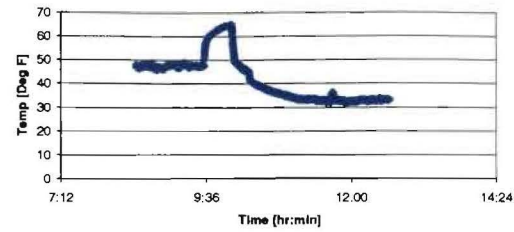
Outside Air Velocity



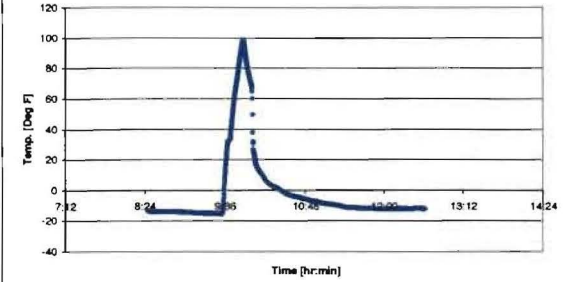
Inside Temp. (Intake)



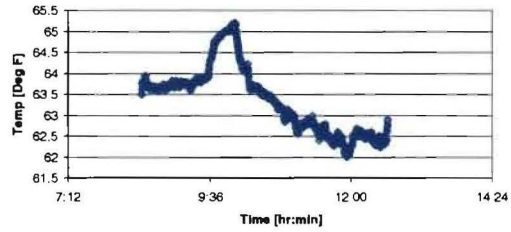
Outside Temp. (HX in)



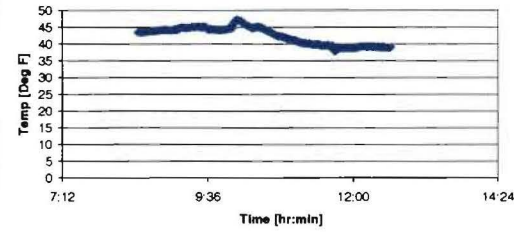
Fluctuations in cooling coil temperature



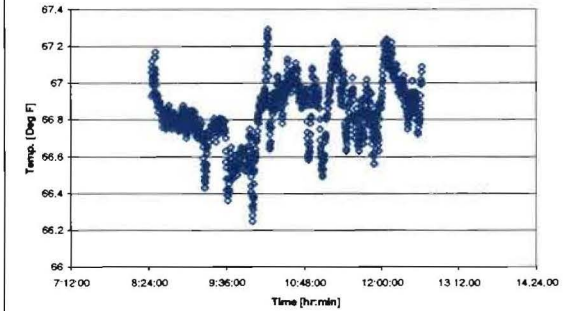
Inside Temp. (exhaust)



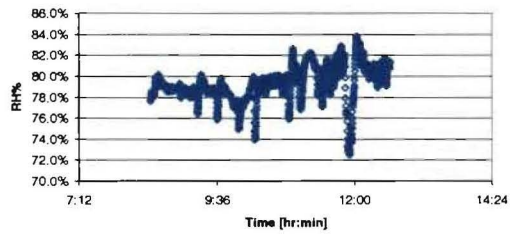
Outside Temp. (HX out)



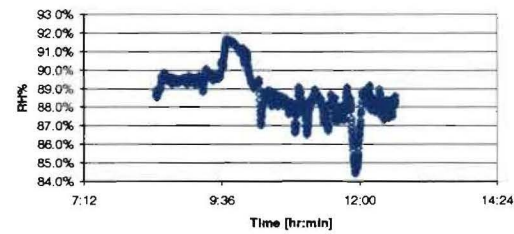
Growth chamber ambient temperature



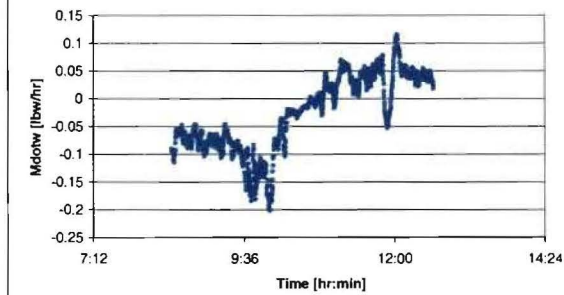
Relative Humidity (In)



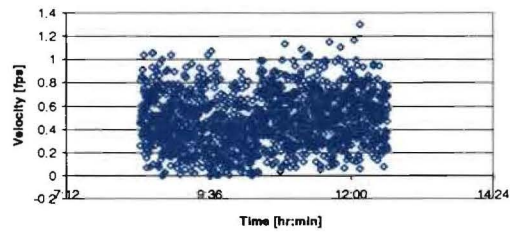
Relative Humidity (Out)



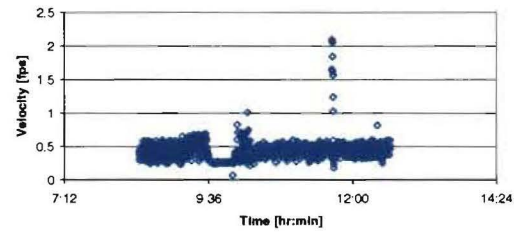
Mass Flow of Water, 2 min. Avg.



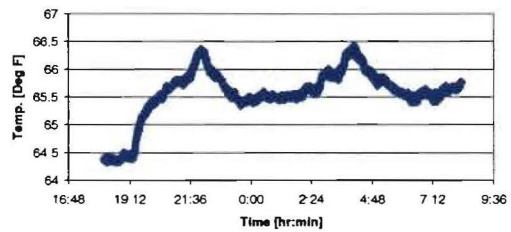
Inside Air Velocity



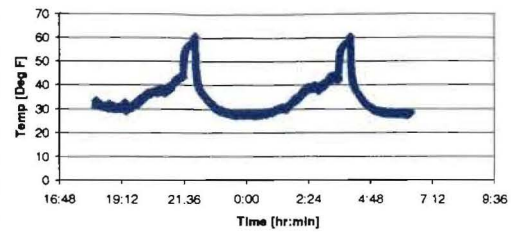
Outside Air Velocity



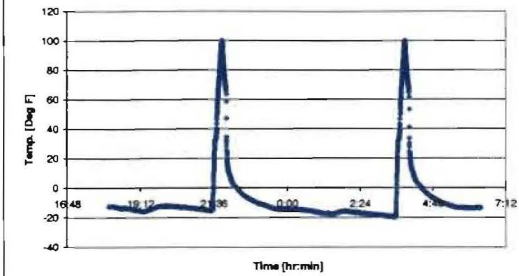
Inside Temp. (intake)



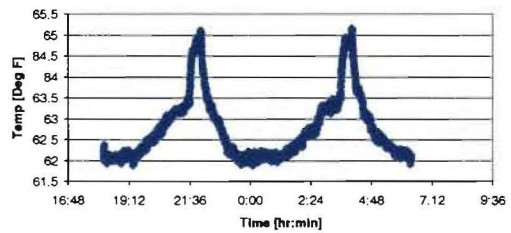
Outside Temp. (HX In)



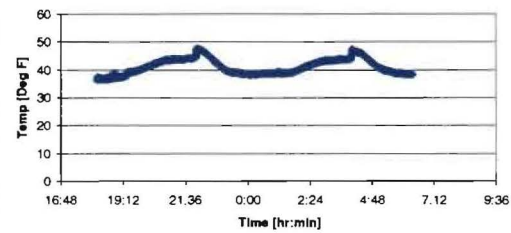
Fluctuations in cooling coil temperature



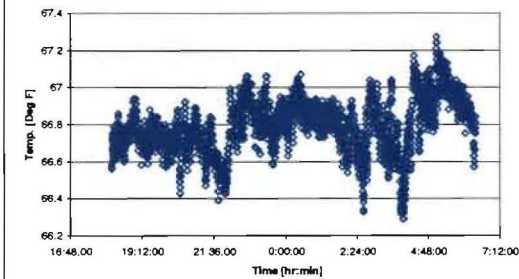
Inside Temp. (exhaust)



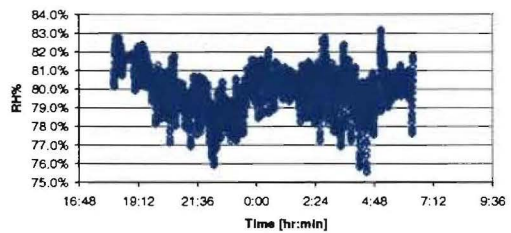
Outside Temp. (HX out)



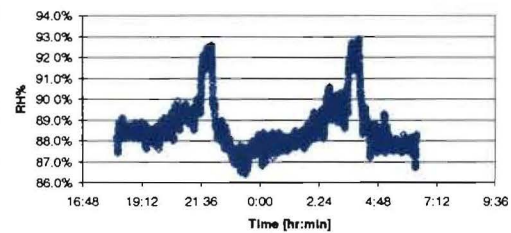
Growth chamber ambient temperature



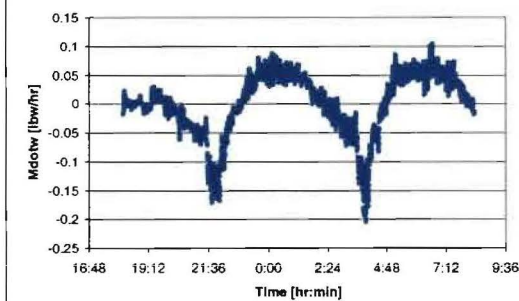
Relative Humidity (In)



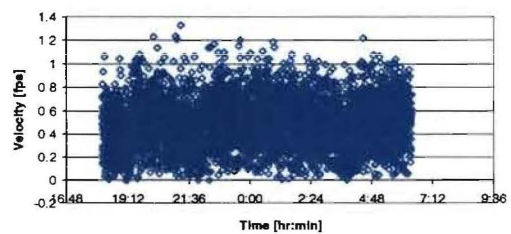
Relative Humidity (Out)



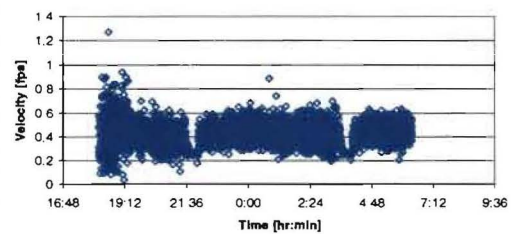
Mass Flow of Water, 2 min. Avg.



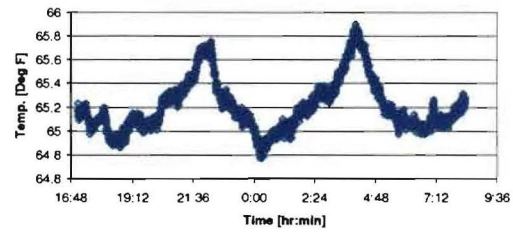
Inside Air Velocity



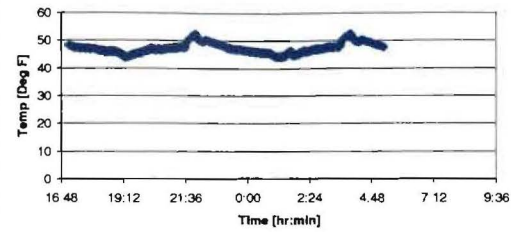
Outside Air Velocity



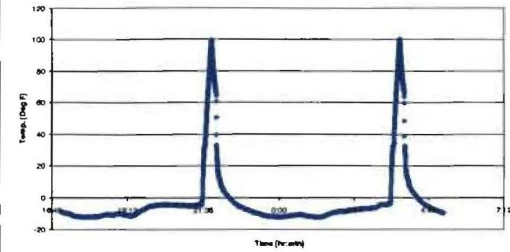
Inside Temp. (Intake)



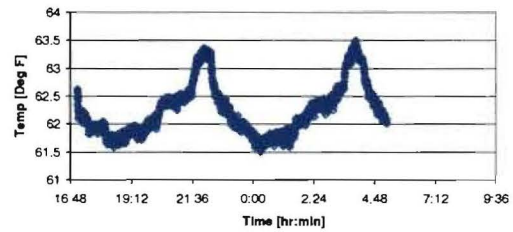
Outside Temp. (HX in)



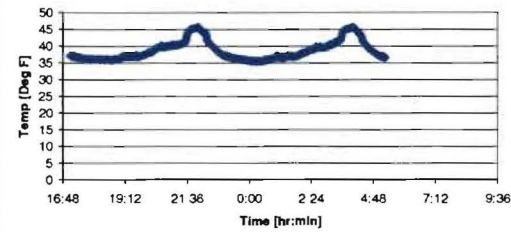
Fluctuations in cooling coil temperature



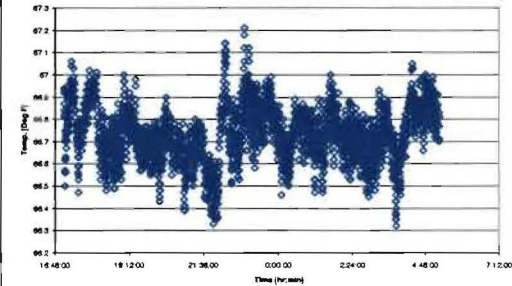
Inside Temp. (exhaust)



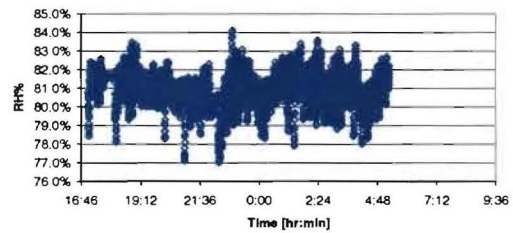
Outside Temp. (HX out)



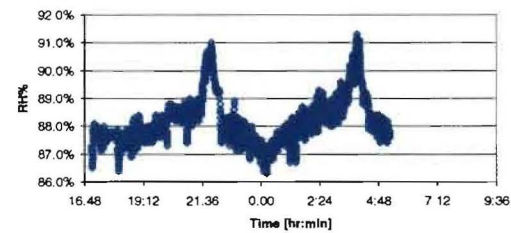
Growth chamber ambient temperature



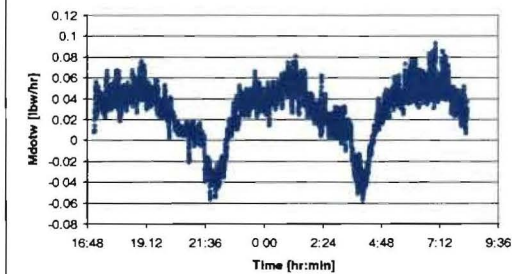
Relative Humidity (In)



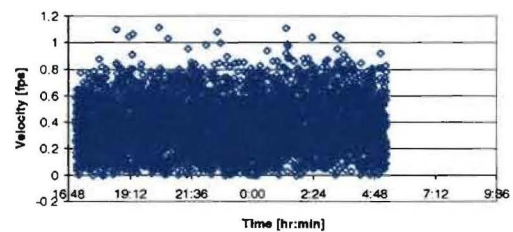
Relative Humidity (Out)



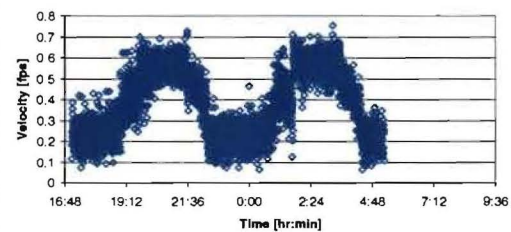
Mass Flow of Water, 2 min. Avg.



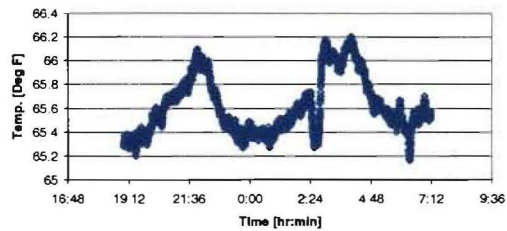
Inside Air Velocity



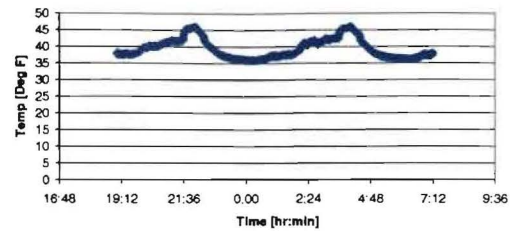
Outside Air Velocity



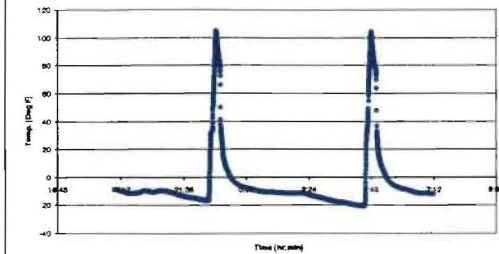
Inside Temp. (intake)



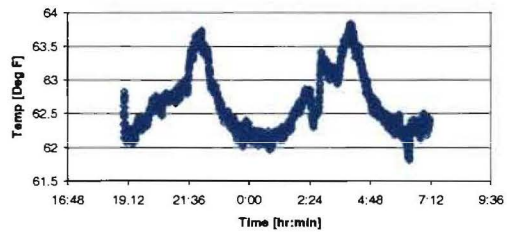
Outside Temp. (HX in)



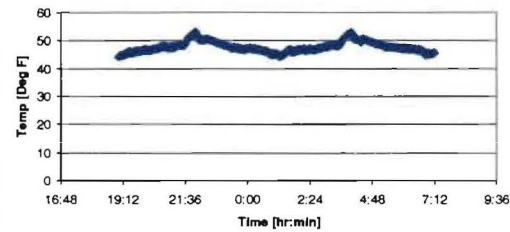
Fluctuations in cooling coil temperature



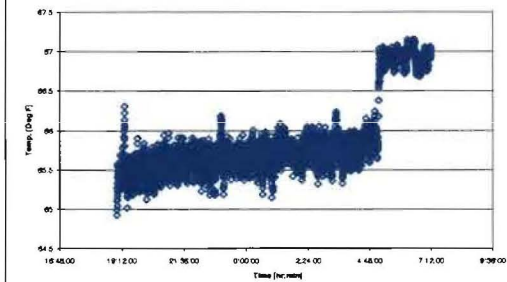
Inside Temp. (exhaust)



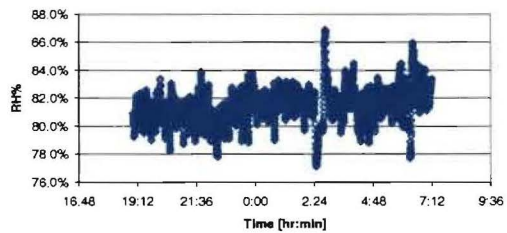
Outside Temp. (HX out)



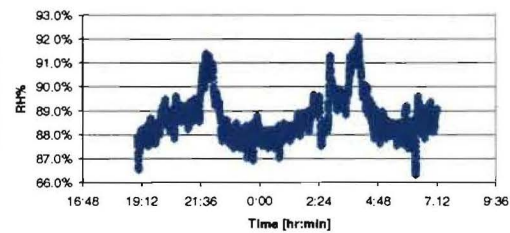
Growth chamber ambient temperature



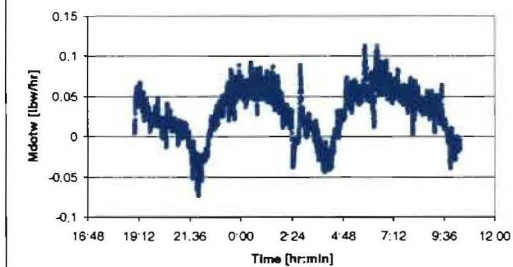
Relative Humidity (In)



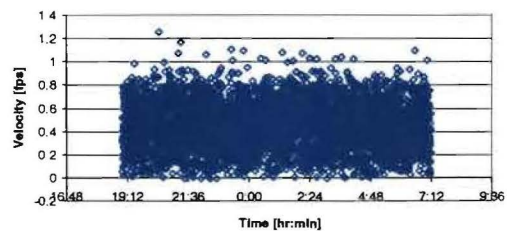
Relative Humidity (Out)



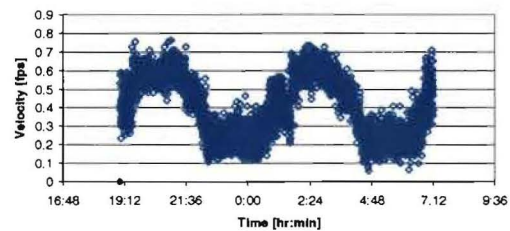
Mass Flow of Water, 2 min. Avg.



Inside Air Velocity



Outside Air Velocity



Appendix C

Description of equipment

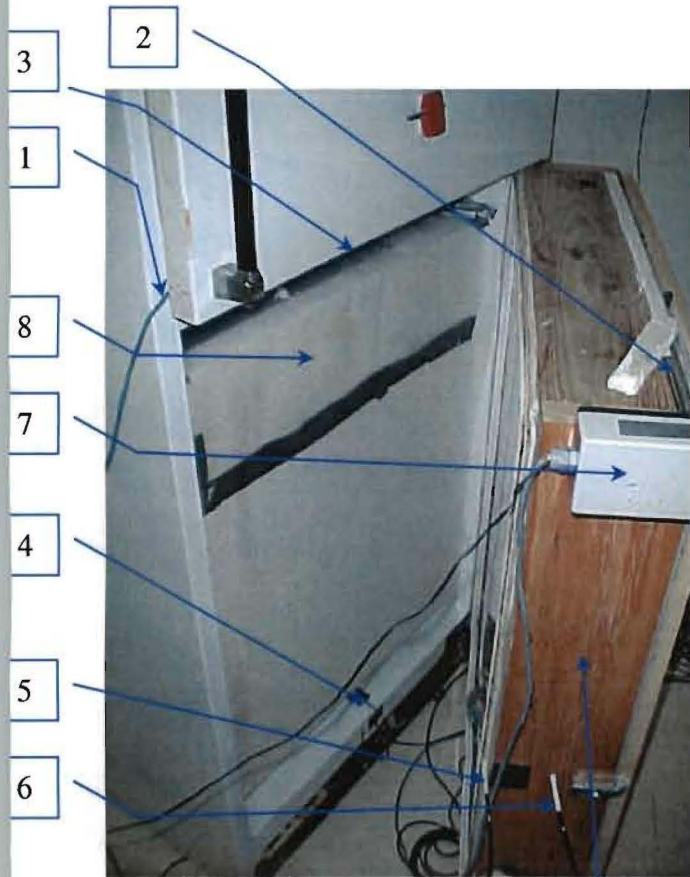


Figure 1: CEDU Prototype



Figure 5: CEDU Prototype (front)

1 – 4 & 14) Type-T thermocouple

5 & 6) Air velocity transducer (TSI, Inc., 8455-12, 0-5V)

7 & 9) Relative humidity transducer (Rotronic, H3V-200SF-005-N, 0-100%RH = 0-5V)

8) Polystyrene insulation, “flow restrictor”

10) Refrigerator fan (Dayton 55CFM Axial Fan, 4C548B, 115V, 0.083A, 6 Watt)

11) Intake fan (Rotron, Inc., Sentinel, Model 747, 14 Watt)

12) Wood & polyethylene door, 6” wide, sealed with 3/16” x 3/8” weather stripping

13) JC Penny refrigerator (Model 867 0220 72 008, 8 Amp, 115) Donated by M. Brugger, Ph.D.



15) Sloping gutter (28" galvanized steel)

16) Intake manifold (2" dia. PVC)

Figure 6: Intake/gutter assembly

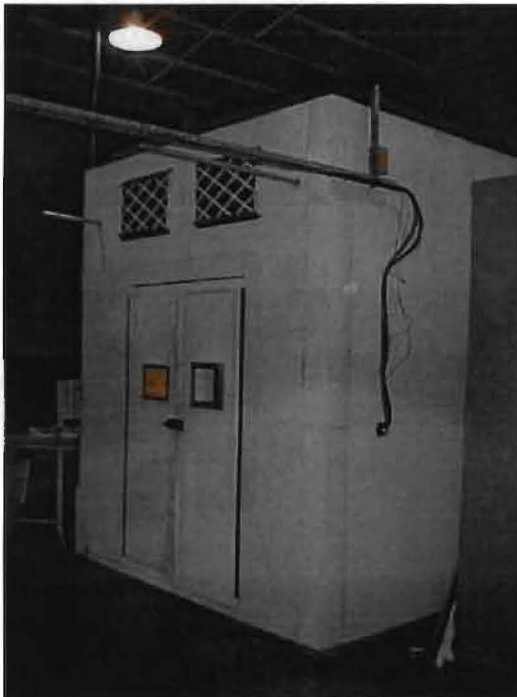


Figure 8: Environmental Growth Chamber.
OARDC, Wooster, OH



Figure 7: Campbell Scientific, CS23x data logger



Figure 9: Sealed collection Beaker

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4. LePoidevin, R.E., D.R. Mears, W.J. Roberts. 1981. A prototype heat exchanger for humidity control in greenhouses. Transactions of the ASAE, Paper No. 81-4526.
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